AD-A050 279 ADVISORY GROUP FOR AEROSPACE RESEARCH AND DEVELOPMENT--ETC F/G 1/3
AN INTRODUCTION TO THE PROBLEM OF DYNAMIC STRUCTURAL DAMPING,(U)
JAN 78 P SANTINI, A CASTELLANI, A NAPPI

UNCLASSIFIED AGARD-R-663 NL

| OF |

AD
A050279

END
DATE
FILMED
3-78
DDC

5 14

AGARD-R-663

AGARD

ADVISORY GROUP FOR AEROSPACE RESEARCH & DEVELOPMENT

7 RUE ANCELLE 92200 NEUILLY SUR SEINE FRANCE

AGARD REPORT No. 663

An Introduction to the Problem of Dynamic Structural Damping

by

Paolo Santini, Antonio Castellani and Alfonso Nappi

12 27p. 11 Jan. 78

NORTH ATLANTIC TREATY ORGANIZATION

DISTRIBUTION AND AVAILABILITY ON BACK COVER

400043

AGARD

NATO OTAN

7 RUE ANCELLE · 92200 NEUILLY-SUR-SEINE

FRANCE

Telephone 745.08.10 · Telex 610176

DISTRIBUTION OF UNCLASSIFIED AGARD PUBLICATIONS

AGARD does NOT hold stocks of AGARD publications at the above address for general distribution. Initial distribution of AGARD publications is made to AGARD Member Nations through the following National Distribution Centres. Further copies are sometimes available from these Centres, but if not may be purchased in Microfiche or Photocopy form from the Purchase Agencies listed below.

NATIONAL DISTRIBUTION CENTRES

BELGIUM
Coordonnateur AGARD – VSL
Etat-Major de la Force Aérienne
Quartier Reine Elisabeth
Rue d'Evere, 1140 Bruxelles

CANADA
Defence Scientific Information Service
Department of National Defence
Ottawa, Ontario K1A OZ2

DENMARK
Danish Defence Research Board
Østerbrogades Kaserne
Copenhagen Ø

FRANCE
O.N.E.R.A. (Direction)
29 Avenue de la Division Leclerc
92 Châtillon sous Bagneux

GERMANY
Zentralstelle für Luft- und Raumfahrt-
dokumentation und -information
Postfach 860880
D-8 München 86

GREECE
Hellenic Armed Forces Command
D Branch, Athens

ICELAND
Director of Aviation
c/o Flugrad
Reykjavik

ITALY
Aeronautica Militare
Ufficio del Delegato Nazionale all'AGARD
3, Piazzale Adenauer
Roma/EUR

LUXEMBOURG
See Belgium

NETHERLANDS
Netherlands Delegation to AGARD
National Aerospace Laboratory, NLR
P.O. Box 126
Delft

NORWAY
Norwegian Defence Research Establishment
Main Library
P.O. Box 25
N-2007 Kjeller

PORTUGAL
Direccao do Servico de Material
da Forca Aerea
Rua de Escola Politecnica 42
Lisboa
Attn: AGARD National Delegate

TURKEY
Department of Research and Development (ARGE)
Ministry of National Defence, Ankara

UNITED KINGDOM
Defence Research Information Centre
Station Square House
St. Mary Cray
Orpington, Kent BR5 3RE

UNITED STATES
National Aeronautics and Space Administration (NASA),
Langley Field, Virginia 23365
Attn: Report Distribution and Storage Unit

THE UNITED STATES NATIONAL DISTRIBUTION CENTRE (NASA) DOES NOT HOLD STOCKS OF AGARD PUBLICATIONS, AND APPLICATIONS FOR COPIES SHOULD BE MADE DIRECT TO THE NATIONAL TECHNICAL INFORMATION SERVICE (NTIS) AT THE ADDRESS BELOW.

PURCHASE AGENCIES

Microfiche or Photocopy
National Technical
Information Service (NTIS)
5285 Port Royal Road
Springfield
Virginia 22151, USA

Microfiche
Space Documentation Service
European Space Agency
10, rue Mario Nikis
75015 Paris, France

Microfiche
Technology Reports
Centre (DTI)
Station Square House
St. Mary Cray
Orpington, Kent BR5 3RF
England

Requests for microfiche or photocopies of AGARD documents should include the AGARD serial number, title, author or editor, and publication date. Requests to NTIS should include the NASA accession report number. Full bibliographical references and abstracts of AGARD publications are given in the following journals:

Scientific and Technical Aerospace Reports (STAR),
published by NASA Scientific and Technical
Information Facility
Post Office Box 8757
Baltimore/Washington International Airport
Maryland 21240, USA

Government Reports Announcements (GRA),
published by the National Technical
Information Services, Springfield
Virginia 22151, USA

Printed by Technical Editing and Reproduction Ltd
Harford House, 7–9 Charlotte St, London W1P 1HD

ISBN 92-835-1268-5

(5)

AGARD-R-663

NORTH ATLANTIC TREATY ORGANIZATION

ADVISORY GROUP FOR AEROSPACE RESEARCH AND DEVELOPMENT

(ORGANISATION DU TRAITE DE L'ATLANTIQUE NORD)

AGARD Report No.663

AN INTRODUCTION TO THE PROBLEM OF DYNAMIC STRUCTURAL DAMPING

by

P.Santini, A.Castellani and A.Nappi
Istituto di Tecnologia
Aerospaziale dell'Università di Roma
Via Eudossiana 16
00184 Roma, Italy

Papers presented at the 45th Structures and Materials Panel Meeting, Voss (Norway) September 1977.

THE MISSION OF AGARD

The mission of AGARD is to bring together the leading personalities of the NATO nations in the fields of science and technology relating to aerospace for the following purposes:

– Exchanging of scientific and technical information;

– Continuously stimulating advances in the aerospace sciences relevant to strengthening the common defence posture;

– Improving the co-operation among member nations in aerospace research and development;

– Providing scientific and technical advice and assistance to the North Atlantic Military Committee in the field of aerospace research and development;

– Rendering scientific and technical assistance, as requested, to other NATO bodies and to member nations in connection with research and development problems in the aerospace field;

– Providing assistance to member nations for the purpose of increasing their scientific and technical potential;

– Recommending effective ways for the member nations to use their research and development capabilities for the common benefit of the NATO community.

The highest authority within AGARD is the National Delegates Board consisting of officially appointed senior representatives from each member nation. The mission of AGARD is carried out through the Panels which are composed of experts appointed by the National Delegates, the Consultant and Exchange Program and the Aerospace Applications Studies Program. The results of AGARD work are reported to the member nations and the NATO Authorities through the AGARD series of publications of which this is one.

Participation in AGARD activities is by invitation only and is normally limited to citizens of the NATO nations.

The content of this publication has been reproduced directly from material supplied by AGARD or the authors.

Published January 1978

ISBN 92-835-1268-5

Printed by Technical Editing and Reproduction Ltd
Harford House, 7–9 Charlotte St, London W1P 1HD

PREFACE

In the design of aerospace structures, damping plays a fundamental role. Lack of knowledge of the damping ratio can result in either serious damage to the structure or over-dimensioned structure. The problem of determination of damping ratios is not yet resolved.

This Report indicates some fundamental aspects of the problem:

- the fields where damping is crucial,
- the types of structure involved,
- materials,
- mathematical simulation,
- test methods.

A bibliographic survey of numerical values completes the Report.

P.SANTINI
Chairman, Sub-Committee on
Structural Damping

CONTENTS

CURRENT PROBLEMS IN THE AREA OF DYNAMIC DAMPING

by

Paolo SANTINI

University of Rome, Rome, Italy

ABSTRACT

Some of the major topics in the area of dynamic damping are described. A list of typical problems where damping is of primary importance is listed. Typical structural components are considered, and a brief account on the effect of materials is given. Mathematical models and intermodal coupling is also examined, and the extreme difficulty to have reasonably accurate informations from them is emphasized. Possible philosophies of ground tests and flight tests are finally discussed.

1.– INTRODUCTION

Dear colleagues, as you all will remember, an ad-hoc group was appointed at our Lisbon meeting, last April, with the aim of *investigating possible actions in the field of "Dynamic Structural Damping" as applied to Aerospace Structures*. The importance of the subject was recognized by the SMP–TPC, and several panel members declared their interest for the subject. During the past six months (which were actually five months, or four months, if one considers August as a dead month) several contacts were held between the participants in the new activity. As a result, two papers will now be read.

The aim of my presentation is to try to define some possible lines of interest in our future work, and to sketch a map of the main goals we should follow before our work expires.

It is known that knowledge of dynamic structural damping is of high interest in all kinds of aerospace structures: aircrafts; missiles, rockets, spacecrafts. However, there are significant differences especially from a practical viewpoint, between the different problems, so that a simplified general approach could not be satisfactory.

The purpose of the following discussion is to investigate about the following items:

(1) Aerospace problems where damping is a crucial point
(2) Types of structures involved
(3) Materials
(4) Mathematical models
(5) Intermodal coupling
(6) Methods of testing
(7) Flight tests

The list is far from being complete; it is intended merely as a basis for the discussion. Any possible implementation or suggestion from panel members or experts, now or later, will be welcomed.

2.– AEROSPACE PROBLEMS WHERE DAMPING IS A CRUCIAL POINT

a) Dynamic response near resonance. The well known linear damped oscillation:

(1) $$mx'' + 2kx' + Kx = F(t)$$

can be written:

(2) $$x'' + 2\zeta\omega x' + \omega^2 x = F(t)/m$$

As $F(t)$ is periodical $F(t) = F_o e^{j\Omega t}$ the steady response amplitude is:

(3) $$F_R = \frac{F_o}{1 - \sigma^2 + 2j\zeta\sigma}$$

where $\sigma = \frac{\Omega}{\omega}$. It is seen that, as $\sigma \neq 1$, i.e., far from resonance, consideration of ζ involves small changes (in amplitude and phase) however, in the vicinity of resonance peaks, as accurate as the evaluation of other quantities may be, a good knowledge of ζ is a dominant factors in the estimation of peak response. In any kind of structure, this is a key-point. I will simply quote some specific points: response of aircraft elements when subjected to steady periodical loads; ceparation of a satellite from the thrusting missile, for which specifications are given of g–response vs. frequency; acoustic vibrations of missile components in the transonic range.

b) POGO – This is a typical instability problem arising in rockets from coupliing between structure, *fuel feedlines, engine*. Several descriptions of the phenomenon are available in technical literature [1] [2] [3]. In any case, theoretical analysis leads to the determination of a critical damping, which is varying with time corresponding to the variation of axial structural frequency and to the variation of coupling with time other parts of the system. Critical dampling ζ_c can be obtained with a reasonable degree of accuracy, and the requirement is that structural damping ζ_s must be greater than ζ_c. Along the flight time there are intervals (A,B, in Fig.1) where such a condition is not satisfied; but, due to the poor knowledge of ζ_s, intervals of instability are very difficult to be appreciated, and could be strongly increased or reduced with respect to the nominal values. Practically no rocket is free from this phenomenon: and the design philosophy reduces to two possible solutions: (i) to require that critical damping be extremely low (or even negative), as, f. i., in Space Shuttle, where a conservative value of $\zeta_c = 0{,}002$ (with respect to the empirical value $\zeta_s = 0{,}01$) is prescribed; this can be obtained also with special PSD (POGO suppression devices) which constitute, in every case, a penalty for the system; (ii) accept a small phase of instability that can rapidly be overcome, (generally centered around the middle phase of the flight). In both cases a precise definition of ζ_s is vital.

c) Supercritical Aeroelastic Vibrations. It is well known, since long that in aeroelastic phenomena, e. g., vibrations of panels at dynamic pressure q greater than a critical value q_c, a limit cycle is established, where the amplitude A is depending on the balance of a cycle average of aerodynamic forces (input) and of dynamic damping.

The amplitude of the limit cycle is directly related to the value of ζ (and is, as known, independent of initial conditions); it is of primary importance for design purposes (maximum allowable stress, fatigue, etc.). Note also that the value of the critical dynamic pressure is not strongly dependent on the damping itself.

d) Loss of stability in spinning spacecrafts. It is known that a spinning top with no damping has two axes of stability (corresponding to the maximum and to the minimum moment of inertia). However, if internal friction, including structural damping, is considered, it is easy to see that the only stable axis is that of minimum I. A more important phenomenon arises in the so-called dual spinner S/C (such as, f. i., INTELSAT IV), a formula widely used in communications satellites. Here stability is ensured by the circumstance that the energy losses in the moving part be greater than the corresponding losses in the steady part. Such losses are, at least partially, those arising from dynamic damping, whose knowledge proves, once more, to be a key factor for stability prediction.

3.– TYPES OF STRUCTURES INVOLVED

As we have seen, practically, all components of aerospace structures are involved in problems of dynamic damping. For each of them a modal description leads, for a single mode, to an equation of the type (1). It would be of highest interest to obtain informations on typical values of ζ to be used in each of the analyses described under Art. 2. It is generally agreed, in fact, that the values of ζ are depending not so much from the "structure" itself, as from the "type" of structures. It is known, f. i., that for variation modes of large boosters, a typical value is $\zeta = 0{,}01$; although even small variations around such reference value may have significant influence on specific phenomena (e.g., POGO) a starting point is extremely useful. We may quote some typical parts for which existing date should be collected:

(i) aircraft components
(ii) aircraft controls
(iii) liquid rockets tanks
(iv) rockets feedlines
(v) spacecraft structures
(vi) spacecraft control systems.

Again, the above list is far from being complete, and any suggestion will be highly appreciated.

4.– MATERIALS

The effect of materials on structural damping is extremely obvious and important. Here, by the term material we denote not the chemical composition only, but also the structural "type" (e.g., composites, sandwichs)

Here again we quote some possible items:

(i) metals
(ii) nonmetallic, or partially metallic materials
(iii) sandwich structures
(iv) composites
(v) honeycombs

For (iii), (iv), (v), dependence of DSD from the process of fabrication should be considered. Also, the effect of anisotropy, nonhomogeneity of materials, variation with temperature might prove to be good points. A more ambitious goal could be gathering sufficient informations for the "design" of a specifically required amount of damping. This is not science fiction, since it has already been done, although in a very rudimentary way, by using honeycomb panels to achieve a high degree

of damping to counteract POGO effects. In such cases, of course, due consideration should be given to the material specific weight.

5.– MATHEMATICAL MODELS

Structural damping results from superposition (and possibly from interaction) between internal viscosity of the material and relative motions of connected parts of the structure. In the first area we should also include effect of creep, plasticity, etc.

(i) The final goal is to obtain sufficiently reliable informations on the physical phenomenon in such a way as to be able to describe it with a mathematical formula. Several attempts have been done in the past for this purpose, but all the mathematical models for the materials have proved to be very inaccurate. This is due, of course, to the tremendous difficulty of the task, as it is for all problems of solid state physics. However, the task should consist of two phases: 1) to construct a reasonable model for the internal behaviour of the material, depending on a certain number of unknown parameters; (ii) try to adjust the values of such parameters to fit with the experimental results. In particular, a clear formulation of item (ii) may provide a better knowledge of nonlinear damping which is sometimes associated with important phenomena ignored in the linear theory (e. g., loss of stability in the vicinity of very small linear damping).

(ii) Similar considerations apply to damping due to the interface between different parts of a complex structure (e.g., bolts rivets, connections, misalignments, relative motions, etc.). The task is even more difficult in this case, on account of the random character rivets of the phenomenon, so that a mathematical model can always be very questionable. The problem is however of primary importance, on account of two basic facts: (i) interface damping is in general the prevailing fraction of total damping; (ii) nonlinear effects (e. g., dependence from vibration amplitudes and frequency should not be ignored).

6.– INTERMODAL COUPLING

Eq. (1) is oversimplified. As a matter of fact, the real equation of the structure reads:

$$\mathcal{L}[W] + I[W] + D[W] = 0 \tag{4}$$

where, $\mathcal{L}, I, D$ are structural, inertial and damping operators applied to the displacement vector W. By using the well-known modal approach,

$$W = \sum_{1}^{\infty}{}_r\, w_r(t)\, A_r(P)$$

Eq. (4) yields for each mode:

$$M_r \frac{d^2 w_r}{dt^2} + K_r w_r + \int_B A_r D[w]\, dB = 0 \tag{5}$$

Eq. (1) would imply that the operator $D[w]$ is not only linearly related to the w, but is also diagonal. Even if the linear assumption is retained, a more correct and refined version of Eq. (1) in the free case would be:

$$\frac{d^2 w_r}{dt^2} + 2\omega_r \sum_{1}^{\infty}{}_s\, \zeta_{rs} \frac{dw_s}{dt} + \omega_r^2 w_r = 0 \tag{6}$$

Intermodal coupling arising from nondiagonal terms of (6) is very difficult to determine, and the situation is aggravated by anisotropy, nonhomogeneity, temperature variations, etc. The difficulty of the task arises mainly from the very great number of nondiagonal ζ_{rs} that should be considered. Evaluation of their effect on the dynamical behavior can sometimes be roughly estimated globally: for example, in POGO analysis, modal coupling may increase critical damping up to a value 20% greater than the nominal one.

7.– METHODS OF TESTING

All the above circumstances clearly indicate that experimental validation of the results is of vital importance. No safe prediction of the phenomena listed under Art. 2 can be achieved without relying on such experimental validity; although, as said, mathematical models and first approximation or empirical values can be of great help at the beginning of a design.

The general principle of tests is based on two methods. The first method is to observe and measure vibration deecay (Fig. 3): this principle is valid for high damping, and for small structures. For large structures, it is more desirable to measure several points along the resonance peaks, fitting the results with the model described by Eq. 3.

In any case, a first question arises, related to the dilemma reduced scale vs. full scale. There is no doubt the latter is better, but it is sometimes impossible (e. g., the case of large boosters, for which the size of the experimental facilities would be not feasible). Results obtained with the reduced scale technique should be used very carefully. An intermediate technique that is sometimes used is to have a full scale model of the portion of the structure where the mode under concern is considered to be more important, simulating the remaining portion with dummy equivalent masses.

Finally, the role of flight tests should be considered. Such tests, in general, cannot provide direct informations on the

specific values of ζ ; they simply provide global informations, to be used for the specific problem under concern; but they can be "stored" for further similar situations. For example, referring again to the POGO problem, Fig. 1, flight tests make it possible to have an idea of the structural damping by observing the phases of instability and by associating the results to the values of critical damping which, as said, can be determined with a good degree of confidence through adequate mathematical results. It is obvious that the main goals of such a test could not be the evaluation of ζ ; however, the informations gathered can be used for design of future similar rockets.

8.– CONCLUSIONS

Some of the major current problems in the area of dynamic damping have been listed. They essentially refer to the type of problem, structure, material, involved. Comparison of theoretical results with experimental results has proved to be essential, even with the limitations associated with any experimental technique.

It is, the writer's opinion that AGARD can make a good work in this area, mainly in the collection and critical evaluation, and comparison of existing and future results. Even a partial, or very partial, accomplishment of the goals listed above can give a significant help to the aerospace structural designers of NATO countries.

SYMBOLS

k	damping coefficient
j	$\sqrt{-1}$
m	oscillator mass
q	dynamic pressure
q_c	critical value of q
t	time
x	oscillator coordinate
$A_r(P)$	r.th mode.
B	body of the structure
D	damping operator
F	forcing input
F_O	amplitude of F
F_R	response amplitude
J	inertia operator
K	spring constant
K_r	equivalent stiffness of r.th mode
$\mathcal{L}$	structural operator
M_r	equivalent mass of r.th mode
W	elastic displacement vector
W_r	amplitude of r. the mode
ζ	reduced damping
ζ_c	critical damping
ζ_s	structural damping
ζ_{rs}	intermodal damping
$\sigma = \frac{\omega}{\Omega}$	
ω	eigenfrequency
Ω	forcing frequency
P	generic point of the structure

BIBLIOGRAPHY

1. S. Rubin, *Longitudinal Instability of Liquid Rockets Due to Propulsion Feedback*, (POGO) J. Spacedrafts and Rockets, Vol. 3, n. 8, Aug. 1966.

2. R.C. Seidel, *Space Shuttle POGO Active Controller Design Using Frequency Domain Optimization*. NASA TM - X - 3368, April 1976.

3. M. Barrère, J. Bouttes, J.J. Dordain, *Incidenza dei fenomeni di combustione sull'effetto POGO*, l'Aerotecnica, Missili e Spazio, Vol. 52, n. 2, 1973.

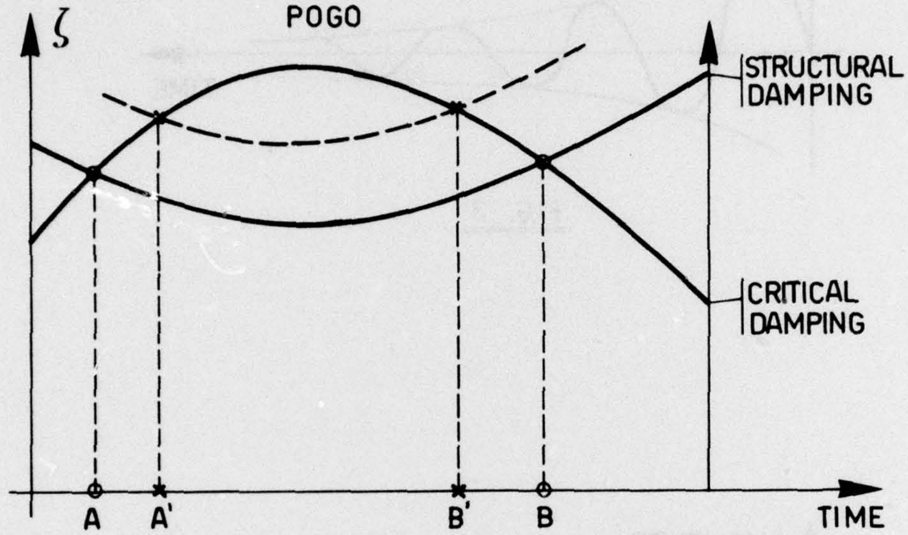

FIG. 1

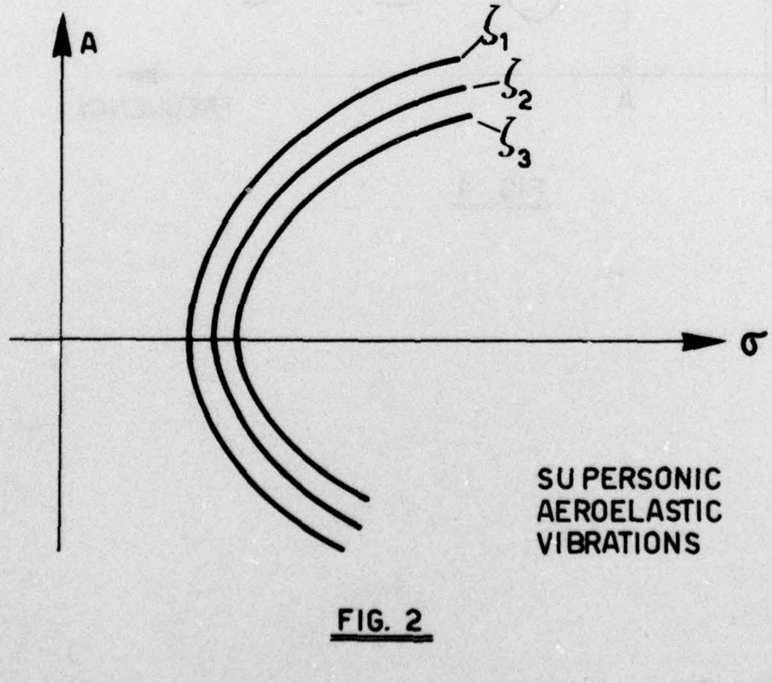

FIG. 2

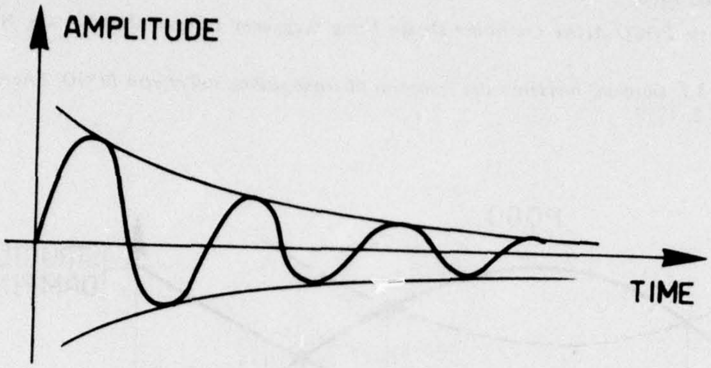

FIG. 3

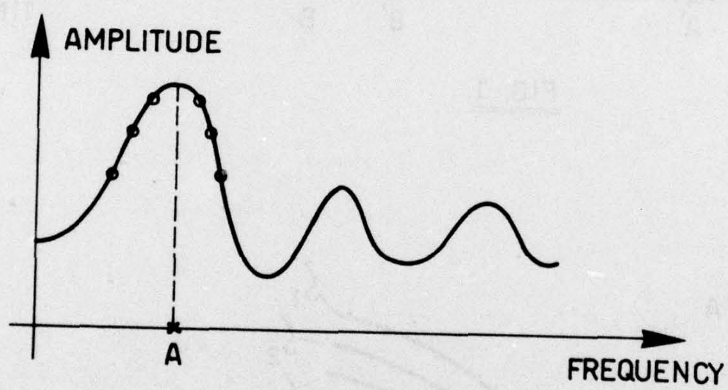

FIG. 4

SOME ASPECTS OF DYNAMIC STRUCTURAL DAMPING PREDICTION

by

Antonio Castellani
Alfonso Nappi

Istituto di Tecnologia Aerospaziale
dell'Università di Roma
Via Eudossiana, 16 – 00184 Roma – Italy

SUMMARY

In the present paper a few aspects concerning dynamic damping prediction for aerospace structures are reported. The problem is of fundamental importance, since the success of a design depends on the knowledge of damping; as a matter of fact the prevision of a damping ratio which is too high will overburden the structure; otherwise serious damage may occur or the design may even be deeply compromised.

Particular attention is given to damping capacities of materials since they are the first source of dynamic structural damping for structures subject to time dependent or shock loading.

This work should represent an introduction to the solution of problems connected with damping prediction and an analysis of some methods commonly used is carried out.

At last a survey of experimental data concerning European satellites is briefly discussed.

INTRODUCTION

In aircraft, rocket and spacecraft design the knowledge of structural damping is one of the preminent items, as flying structures are subject to time-dependent excitation and vibrations are often responsible for very serious damages both to the structure itself and to people and instruments. Acoustic excitation, Pogo effect and vibrations due to the propulsive systems are just an example of time-dependent forces acting on aerospace structures.

Damping leads to energy dissipation when materials are deformed: if a force were applied to a completely elastic material, this would start oscillating about its neutral position when that force is removed, owing to the energy stored in the material; on the contrary, if the force were applied to a completely plastic material, all the work would be dissipated and when the force is removed no more displacements would take place. The first one is known as undamped system, while the second one is a completely damped system. Obviously any material is neither perfectly plastic nor elastic: only a certain amount of energy can be stored or dissipated and the ratio between the plasticity and elasticity of a material describes its behavior and is usually referred to as damping factor of the material itself[1].

Obviously in complex systems, such as aerospace structures, damping is not due only to the plastic characteristics of materials: friction damping at joints and damping introduced by the environment are also to be accounted for. Therefore a damping ratio which includes all the forms of damping is generally introduced, but unfortunately its prediction is really a hard job and represents one of the main difficulties in structural design, because of the several damping mechanisms involved and of the poor results achieved in the attempt of synthesizing elementary models into complex structural systems.

This paper has been written with the aim of giving a little contribution to introduce the problem of dynamic structural damping and to summarize some important results obtained during the last decade.

The main expectation is that the present work, carried out in a rather short period, can be extended and improved, in the next future, by adding more and more data, since an accurate collection and discussion of experimental results appears, at present, as a possible way for predicting damping ratios in structures similar to previous ones, for achieving designers' higher confidence and for lightening structures by reducing uncertainties; another object to be evaluated is obviously the possibility of reducing the number and cost of experimental tests in future structures. On the other hand it is easy to forecast more and more studies and research activities concerning this subject, as it can be seen by observing Fig. 1, where the number of publications on damping is reported up to the 60's according to Ref. 2 (the dot line representing the cumulative number per year).

Damping nomenclature

Although it is well known, it might be worthwhile underlining the enormous importance of the damping factor at resonance, since the amplitude of vibrations (and the value of the acceleration applied to the system) at this particular frequency depends only upon that factor.

As a matter of fact, if an elementary single degree of freedom system is considered, its motion is governed by the second order equation:

$$m\ddot{q} + c\dot{q} + kq = F \sin \omega t$$

or:

$$\ddot{q} + 2\zeta\omega_R\dot{q} + \omega_R^2 q = A \sin \omega t$$

where:

$$\omega_R = \sqrt{k/m} \quad \text{(resonant frequency)}$$

$$A = F/m$$

$$\zeta = c/c_{crit} = c/(2\sqrt{mk})$$

since c_{crit} is the minimum value of c which leads to a non-oscillatory response. The solution of the equation is:

$$q = \frac{A}{\sqrt{(\omega_R^2 - \omega^2)^2 + 4\zeta^2 \omega_R^2 \omega^2}}$$

and, at resonance:

$$q_R = A/(2\zeta \omega_R \omega)$$

Hence an error in the prediction of ζ (which shall be called "damping ratio") would obviously cause the response to be higher or smaller than expected: in the first case serious damages may occur; otherwise useless weight would have been added to the structure.

The variable representing the evolution of a given quantity (such as pressure in combustion chambers of liquid propellent rockets, modal characteristic of structures, flow in ducts, and so on) can be expressed as follows:

$$\Phi(t) = \Phi_o e^{\zeta\omega t} e^{j\omega t}$$

where Φ is either the pressure, or the modal coordinate, or the flow; moreover:

$\zeta = 0$: stability limit
$\zeta > 0$: instability
$\zeta < 0$: stability

If ζ_s is the structural generalized damping ratio for a given mode and ζ_c the critical damping ratio, the following relation can be written:

$$\zeta = \zeta_c - \zeta_s$$

Thus ζ_s must be as high as possible, but it is approximately 1% in aerospace structures; also ζ_c may be of the same order (e.g.: Diamant launcher) so that a little uncertainty may lead to instability. Sometimes it is possible to diminish ζ_c, but this reduction is likely to increase ζ_c in close modes. In such cases only an increase of ζ_s can be realized, as it will be discussed later on.

The damping ratio is then an index of the system ability of dissipating part of the vibration energy and its exact knowledge would be fundamental for structural design.

In Fig. 2a the great influence of the hysteretic damping upon the response of a structure is shown, while in Fig. 2b the maximum deflection of a beam is reported as a function of ζ.

Experimental measurement of damping

The knowledge of damping properties is absolutely necessary for a reliable prediction of the loads applied to a structure subject to time-dependent forces and vibrations.

It is well known that damping depends on joints, materials, equipment and every other structural component: thus theoretical estimation of overall damping is extremely difficult and so far experimental data have represented the only source of reliable informations; as to the existing data on single components, they are not sufficient to determine overall damping starting from the knowledge of individual contributions since repeated tests at different stages of aerospace structure development are really lacking.

Detailed descriptions of the methods used to predict damping can be found in the literature so that only a quick list of the most common techniques is presented here[3]:

Frequency response method:
- half power point method
- phase change method

Measure under transient conditions:
- starting transient and decay transient

Energy dissipation method.

When a sinusoidal test is carried out a correct value of damping can be determined by means of dynamic calculations; in actual fact the exact value of the damping ratio is the one which leads to the same response as that measured experimentally.

DAMPING OF MATERIALS

Damping capacity of materials is highly important in reducing the danger of failure due to fatigue when structures are subject to cyclic or shock loading and in reducing the vibrations transmitted to adjacent components. There is a big variety in damping capacities of materials and no relation has been found with mechanical properties, although damping should be some complex function of Young's modulus, elastic limit, number and distribution of voids, and so on; as a rule it can be said that high damping capacity is associated with high degree of plasticity, but there are some exceptions: for instance gray cast iron (a very brittle material) has outstanding damping properties thanks to the graphite flakes of its structure.

When materials are subject to external loads the nature of the applied forces may affect damping properties; obviously the previous stress history and the environment have great influence upon these properties. Sinusoidal excitation has been studied to a large extent, while informations concerning random forces are rather lacking, so that experimental work on damping properties of materials under random vibrations is highly recommended: moreover in the last period the effect of these vibrations has become particularly significant in connection with fatigue and acoustical excitation.

It is well known that cyclic load deformation does not lead to a single valued function $\sigma = \sigma(\epsilon)$, but creates a hysteretic loop. The area within this loop is proportional to the energy absorbed and this energy dissipation can be defined as "damping" or "mechanical damping" or "dynamic damping", which concerns also energy dissipation at interfaces or joints, but excludes energy loss due to electro-hydro-mechanical interactions[2].

Damping in uniform materials is due to movements at microscopic level, interactions among molecular forces, flow phenomena and general viscoelastic effects.

Composites, as it is well known, represent a particular kind of materials with the possibility of being "tailored" so as to improve those properties which are highly required: hence a continuous development and diffusion of composites both for aircraft[4] and for spacecraft[5]. Therefore composite materials able to provide high damping can be studied or improved, as well as viscoelastic adhesives producing high damping at high amplitudes.

Damping is produced also by joints, which are characterized either by dry interfaces or lubricated interfaces or adhesively bonded interfaces. Both for dry and lubricated interfaces two kinds of motion are possible: motion normal to the interfaces and relative motion of surfaces in the plane of the interfaces. Obviously higher damping capacity is determined by relative motion (or relative interface shear) obtained with dry interfaces. In this case, however, serious problems arise owing to fretting and corrosion, so that lubricated interfaces have been studied deeply; similarly, the behavior of plastics and any other non-fretting material located at interfaces has been investigated in the attempt of avoiding wear as much as possible.

Unfortunately fretting has not been avoided completely or the amount of dissipated energy is rather low. Interesting results have been accomplished by using adhesive interfaces with a sufficient thickness to allow relative motion between the jointed surfaces; in this case the energy due to the relative motion of the surfaces is absorbed within the adhesive.

In composites each ply can be seen as an adhesive layer and its behavior must be analyzed to determine the composite damping characteristics, which turn out to be a function of the contributions of each single layer.

A perfectly elastic material would be characterized by no rate-dependent effect; rate-dependence means that the stress-strain relation depends on the rate of loading (dP/dt, $d\sigma/dt$) or the rate of straining (dl/dt, $d\epsilon/dt$).

After applying a sinusoidal load to a material, a residual strain will be present, in general, when the load is reduced to zero; the residual strain, however, does not necessarily remain constant with time: it may either go down to zero or to some intermediate value. Something similar may happen when a residual constant load is left: the strain may either go down to the only value that would be related to the applied stress if the material were perfectly elastic or go down to an intermediate value.

Obviously the strain may well happen to depend only on stress, although the function $\sigma = \sigma(\epsilon)$ is not single valued and a loop appears if this function is plotted: however the loop width is zero at zero stress owing to its dependence only on the stress itself: this particular behavior is known as "elastic hysteresis".

Thus the following classification is possible:

Rate–independent, non-recoverable behavior;
Rate–dependent, non-recoverable behavior;
Rate–dependent, recoverable behavior.

Obviously rate-independent, recoverable behavior does not appear to the possible since recovery is a time dependent effect and must be excluded by the rate-independent behavior.

According to the classification just made, it is possible to speak of rate-dependent and rate-independent damping: the first one will depend on loading frequency and will be described by stress-strain equations involving time-derivatives; both kinds of damping may depend on stress amplitude.

Damping-stress relations are generally complex, but sometimes the following expression can be used:

$$D = C\sigma^n$$

where C is the damping energy dissipated when the stress amplitude is equal to one and n is the "damping exponent".

At low amplitudes the so called "quadratic damping" is observed: n is equal to 2 and the hysteretic loop is elliptical; the definition "linear damping" is also used since linear differential equations and linear superposition are normally appropriate.

At higher amplitudes non-linear damping is observed, characterized by non elliptical loops and values of n different from 2.

In rate-independent materials non linear damping is associated with plastic strain and magnetoelasticity[3-6].

The damping properties of materials can be expressed by means of the phase displacement φ between stress and strain, and by introducing the complex Young's modulus:

$$E(\omega) = E(1 + j\zeta)$$

where $\zeta = tg\,\varphi$.

Hooke's law can now be expressed in its most general form:

$$\sigma = A_0\epsilon + A_1\dot{\epsilon} + A_2\ddot{\epsilon} + \ldots$$

and it is possible to write the following relation, which holds for a given frequency:

$$\bar{\sigma} = \bar{E}(\omega)\bar{\epsilon}$$

Great interest has been shown recently in composites and particular attention has been given to the damping properties of sandwich structures.

For instance an attempt has been made to predict these properties from the dynamic characteristics of core and facings, measured

by means of the decay method, but theoretical and experimental results have been quite different[7].

An interesting characteristic of sandwich structures is the following one: the damping ratio tends to be constant up to a certain amplitude, but then it increases rapidly.

The parameters which are of interest in determining sandwich damping have not been studied deeply and in general only beams subject to flexural deformations have been considered; it is possible, however, to make the following remarks:

- aluminium honeycomb sandwiches vibrating in flexural modes have a very small damping ratio (below .15%)
- the main contribution to the damping ratio under the same condition is determined by the adhesive
- the overall damping of an assembled structure made of sandwich components depends mainly on other items, such as joints and air damping[7]

INCREASE OF STRUCTURAL DAMPING BY MEANS OF VISCOELASTIC MATERIALS

The range of damping values for materials of everyday use is schematically reported in Fig. 3[8]. They are summarized in the following table:

Metals	$10^{-4} \div 10^{-3}$
Assembled structures	$10^{-3} \div 10^{-2}$
Alloys	$10^{-2} \div 10^{-2} \cdot 3$
Simple cover	$10^{-2} \div 10^{-1}$
Viscoelastic materials	> 2

Thus viscoelastic materials have damping ratios over 2, but their elastic modulus is below $10^9\ Nm^{-2}$ so that they are not suitable for mechanical constructions; however the viscoelastic covering technique allows to associate the good characteristics of high modulus materials with the damping capacity of viscoelastic materials, so as to reach damping ratio values between 0.01 and 1. In figures 4[8]a and b the damping characteristics of viscoelastic materials are reported as functions of temperature and frequency. It can be observed that the maximum corresponds to the vitreous state transition temperature, in a range of temperatures which is generally small for a given frequency, so that these materials are highly sensitive to temperature.

The increase of the damping ratio can be achieved by means of simple covering (Fig. 5[8]a), where the viscoelastic layer is associated with an elastic layer, or by means of multiple covering (Fig. 5[8]b) with another elastic layer.

In the case of simple covering the dynamic performance of the viscoelastic layer (subject to tension and compression) depends on:

E_v/E_e (ratio between the moduli of the viscoelastic and elastic materials);

t_v/t_e (ratio between the layer thicknesses);

the intrinsic damping ratio of the viscoelastic material.

This technique is particularly suitable for thin structures and high modulus viscoelastic materials are required. It is possible to have damping coefficients between 0.1 and 0.2.

In the case of multiple covering the performance depends on much more parameters and the covering, subject to shear stresses, allows to achieve better results by increasing the energy dissipated in the viscoelastic layer. If the layers are designed properly and the modulus is between 10^6 and $10^9\ Nm^{-2}$, the damping ratio can be as high as 0.1 and 1. Higher values can be obtained by using more layers, one elastic and one viscoelastic alternatively.

The viscoelastic cover method has been used to correct the POGO effect on the Diamant launcher.[8]

MEASUREMENTS PERFORMED ON SUBSTRUCTURES

One of the methods normally used for structural design purposes is to derive damping characteristics from existing structures and to check the new vehicle by full scale tests. This method, however, might prove to be unpractical in the future for big and complex structures, owing to the cost and difficulties involved in full scale tests. The first vehicle which has created a completely new set of problems is probably the Space Shuttle and new methods have been evaluated to accomplish reliable results by means of "reasonable" procedures; accurate investigations have been carried out to synthesize the results obtained from tests on substructures, with the aim of achieving the necessary informations on the whole system without performing tests on the assembled structure. The principal goal has been, at first, to realize a shuttle model consisting of two substructures, to test these substructures, to develop mathematical models both for the single substructures and for the whole system, and to compare the predicted characteristics with those obtained experimentally[9]. One of the difficulties involved in this method is that resonant frequencies of the whole system are different from the ones of the individual substructures so that it is necessary to evaluate the modal damping energy for all the components, to extrapolate their contributions to the total modal energy at the resonant frequencies of the vehicle and to sum all the results. In actual fact the task to be accomplished is to correlate the substructure results.

If $T = m\,\omega_R^2\, q_R^2 / 2$ is the peak steady-state kinetic energy at amplitude q_R (m being the modal mass), it is possible to express the damping energy dissipated per cycle (D_c) as a function of T and q_R / l, l being a characteristic length.

It is not easy, however, to find out the kinetic energy correctly at low frequencies, when amplitudes have those values which are appropriate at high frequencies. In other words an experimental plot can't be obtained easily for certain values of q_R.

Anyhow, if damping forces can be considered to depend linearly on q_R and damping energies to depend linearly on q_R^2,

measurements can be carried out at any amplitude and no problem will arise to find out the value of T correlated to the selected value of q_R.

If the same conditions are met, the modal damping ratio ζ will be measured as well, by means of usual techniques (half band or free decay). Now the damping energy D_C is obtained straight away, since $D_C = 4\pi T\zeta$.

After plotting the points that give the values of D_C as a function of T (q_R/l being held constant) it is possible to draw a curve through these points: such curve will not be only a plot of modal damping energy vs peak modal kinetic energy, but also a plot of total damping energy vs total peak kinetic energy: to tell the truth this is only an assertion that should be proved by tests performed at off-resonant conditions, but such tests are so inaccurate that the evidence of this hypothesis has been achieved through another kind of measurements.

The first step in the system analysis is the determination of natural frequencies, mode shapes and kinetic energies for a certain modal amplitude q_R : all these results can be obtained quite easily by means of usual computational techniques applied to the undamped system.

Let us suppose, now, that the system has been divided in two substructures. When a particular modal amplitude is selected, it is possible to plot the function $D_C = D_C(T, q_R/l)$ for both substructures (as it was pointed out before), without considering, for the time being, the possibility that they actually assume the given spatial shape. The first or the second substructure, however, will have amplitude q_R, while the other component will probably have a smaller maximum amplitude $\bar{q}_R$ at the mode considered; in any case it will be $\bar{q}_R < q_R$ or $q_R = s\,\bar{q}_R$, where s (> 1) is the scale factor obtained when q_R is divided by $\bar{q}$.

After evaluating the kinetic energies of the substructures corresponding to amplitude q_R (T_1 and T_2), after finding the corresponding values of D_C ($D_{C.1}$ and $D_{C.2}$), and after dividing one of the two energies (let it be $D_{C.2}$) by s^2, so as to obtain $\bar{D}_{C.2}$, it will be easy to predict the system damping ratio:

$$\zeta_s = (D_{C.1} + \bar{D}_{C.2})/(4\pi T_s)$$

where T_s is the kinetic energy of the whole system.

Fig. 6 (taken from Ref. 9, which is the source of the informations given in this section up to now) is an example of two curves $D_C = D_C(T, q_R/l)$ obtained for models of the Shuttle booster and orbiter in similar test conditions.

It is possible to account also for joint damping: this is accomplished by determining the properties of the joint damper and the relative velocity across the joint[9].

The results obtained during the study of a simple model of the Space Shuttle were quite satisfactory, but further development appeared to be necessary before applying this method to actual structures. Ref. 10 represents a progression of the dissipative energy approach for predicting damping. A four component, three dimensional model including liquids in the tank has been developed; also joints that simulate the ones studies for the Shuttle have been considered. Then an analytical model has been found for the substructures and they have been combined into a single model for the whole system. After predicting the modal characteristics (with the exception of dynamic damping) from this model and determining the properties of the substructures and of the joints by means of dynamic tests, the modal damping properties of the system have been predicted through the energy method.

The steps leading to the system damping ratio can be summarized as follows[10]:

- Determination of the kinetic energy T_i of each component at amplitude h_i and determination of the strain energy S_i of each connecting link at amplitude d_i;
- Determination of the strain energy U_i of each component at amplitude h_i ($U_i = T_i - S_i$);
- Normalization to $X_0 = 1\ mm$ maximum amplitude so as to obtain $\bar{U}_i$ and $\bar{S}_i$;
- Determination of damping energies $\bar{D}_{hi}$ and $\bar{D}_{di}$ (corresponding to $X_0 = 1\ mm$ deflection) which can be found by means of experimental curves:

$$\bar{D}_{hi} = f(\bar{U}_i) \quad \text{and} \quad \bar{D}_{di} = g(\bar{S}_i);$$

- Putting $H = 1\ mm$ the damping energy levels are adjusted to the proper amplitudes by means of the equations:

$$D^*_{hi} = \bar{D}_{hi}(h_i/H)^2 \quad \text{and} \quad D^*_{di} = \bar{D}_{di}(d_i/H)^2;$$

- Determination of total kinetic energy T_0 and of total damping energy D^*_0 summing all the contributions;
- Determination of damping ratio for a given mode from the equation:

$$\zeta = D^*_0/(4\pi T_0)$$

The values obtained appear to be quite good at higher modes, where the component structural response is dominant; at low frequencies, on the contrary, connecting link damping plays first fiddle and the results are not correct[10].

CORRELATION BETWEEN MODAL DAMPING AND LOCAL DAMPERS[11]

Dynamic structural analysis is often carried out by developing a spring-mass damper model of the structure; the constants necessary to characterize springs and masses are easily found by performing measures on the weight and stiffness of each section. These constants are sufficient to find out the approximate resonant frequencies of the system; for a complete description of this system, however, damping characteristics must be determined.

The procedure discussed here and proposed in Ref. 11 should turn out to be a really valuable help for modifications of a structure. In actual fact this procedure allows to determine the characteristics of local dampers when modal damping is known. Therefore, if the modal damping of a structure has been found experimentally or is assumed thanks to previous experience on similar structures, it is possible to know how local dampers behave in the spring-mass-damper model, so that the

dynamic characteristics of the structure after eventual modifications can be predicted by studying and testing only the modified component.

If a model of n masses, springs and dampers is developed (Fig. 7), the equation of motion for the i-th section is:

$$F_i = m_i \ddot{q}_i + c_i(\dot{q}_i - \dot{q}_{i+1}) + k_i(q_i - q_{i+1}) - c_{i-1}(\dot{q}_{i-1} - \dot{q}_i) - k_i(q_{i-1} - q_i)$$

When the Laplace transformation with zero initial conditions is taken, the equation becomes:

$$[m_i s^2 + (c_{i-1} + c_i)s + k_{i-1} + k_i]\, q_i(s) - (c_i s + k_i)\, q_{i+1}(s) - (c_{i-1} s + k_{i-1})\, q_{i-1}(s) = F_i(s)$$

If a sinusoidal force is applied only to the n-th mass (bottom mass), as it happens with launcher test facilities, and the boundary conditions are:

$c_{i-1} = k_{i-1} = 0$ at the top mass

$q_{n+1} = 0$ at the bottom mass

the system of equations for the whole model is:

$$[B]\{q(s)\} = \{F(s)\}$$

where:

$$B_{11} = m_1 s^2 + c_1 s + k_1$$

$$B_{ii} = m_i s^2 + (c_{i-1} + c_i)s + k_{i-1} + k_i \qquad \text{for } i > 1$$

$$B_{i,i+1} = B_{i+1,i} = -(c_i s + k_i)$$

all other $B_{ij} = 0$

$$F_i(s) = 0 \qquad \text{for } i \neq n$$

By using Cramer's rule, it can be found:

$$\frac{q_1(s)}{F_n(s)} = \frac{(c_1 s + k_1)(c_2 s + k_2)\ldots(c_{n-1}s + k_{n-1})}{a_{2n} s^{2n} + a_{2n-1} s^{2n-1} + \ldots + a_1 s + a_0} \tag{1}$$

Then it can be shown that the values of the coefficients a_{2n-j} are calculated by means of the following formula:

$$a_{2n-j} = \rho_j = -\frac{1}{j}(\rho_{j-1}\gamma_1 + \rho_{j-2}\gamma_2 + \ldots \rho_1 \gamma_{j-1} + \gamma_j)$$

where: $j = 1, 2, \ldots, 2n$

$\gamma_t = \text{trace}\,[A]^t$ for $t = 1, 2, \ldots, 2n$

$$[A] = \begin{bmatrix} [O] & [I] \\ -[k][m]^{-1} & -[c][m]^{-1} \end{bmatrix}$$

Now modal constants must be related to the response of the system. If the damping ratio is about 10% or less, the following expressions are true:

$$c_{eq}/m_{eq} = 2\zeta\,\omega_{R.l} \qquad \text{for } l = 1, 2, \ldots, n$$

$$|q|_p / |q|_{\omega=0} \simeq 1/(2\zeta) \qquad \text{with } |q|_p = \text{peak amplitude}$$

$$\omega^2_{R.l} = k_{eq}/m_{eq}$$

$$|q|_{\omega=0} = |F|/k_{eq}$$

Hence:

$$c_{eq} \simeq |F|/(\omega_{R.l}\,|q|_p) \tag{2}$$

Putting $s = j\omega$, Eq. 1 becomes:

$$\left|\frac{q_1(j\omega)}{F(j\omega)}\right| = \left|\frac{(jc_1\omega + k_1)(jc_2\omega + k_2)\ \ldots\ (jc_{n-1}\omega + k_{n-1})}{[a_{2n}(-\omega^2)^n + a_{2n-2}(-\omega^2)^{n-1} + \ldots + a_0] + j[a_{2n-1}(-\omega^2)^{n-1} + \ldots + a_1]\,\omega}\right|$$

When $\omega = \omega_{R.l}$ the real portion of this equation can be considered equal to zero. This assumption holds when light damping

is present; it should be noticed that if no damping is present the denominator of the equation must consist only of the real part, which must go to zero in order to have an infinite resonance.

When light damping is present the small change of resonant peaks and the small shift in the resonance frequency can be neglected; so the previous equation becomes:

$$\left|\frac{q_l(j\omega_{R.l})}{F(j\omega_{R.l})}\right| = \left|\frac{(jc_1\omega_{R.l}+k_1)(jc_2\omega_{R.l}+k_2)\ \dots\ (jc_{n-1}\omega_{R.l}+k_{n-1})}{j[a_{2n-1}(-\omega^2_{R.l})^{n-1}+a_{2n-3}(-\omega^2_{R.l})^{n-2}+\dots+a_3(-\omega^2_{R.l})+a_1]\,\omega_{R.l}}\right|$$

This equation must now be inverted to determine $|F|/(|q|_p\,\omega_{R.l})$; the inversion can be performed assuming $c_i^2 \ll (k_i^2/\omega^2_{R.l})$, in order to substitute $(c_i s + k_i)$ by k_i. It must be underlined that this substitution can lead to errors in the solutions at high resonant frequencies. With this approximation, however, and by using Eq. 2, it is possible to write:

$$\frac{|F|}{|q|_p\,\omega_{R.l}} = (c_{eq})_{\omega=\omega_{R.l}} = \frac{1}{\prod_{i=1}^{n-1}}\,[a_{2n-1}(-\omega^2_{R.l})^{n-1}+a_{2n-3}(-\omega^2_{R.l})^{n-2}+\dots+a_1]$$

Finally a last approximation is introduced by putting any product $c_i\,c_k$ equal to zero; thus the coefficients a_{2n-j} become linear functions of the local dampers c_i.

Hence:

$$(c_{eq})_{\omega=\omega_{R.l.}} = |a_1 c_1 + a_2 c_2 + \dots + a_n c_n| \quad \text{for the } l\text{-th mode.}$$

Therefore the whole system of equations that will allow to find the values of every c_i can be written as follows:

$$\{c_{eq}\} = |[a]\ \{c\}|$$

where $[a]$ is a matrix $n \times n$, while $\{c\}$ and $\{c_{eq}\}$ are vectors of n terms.

SOME DATA CONCERNING EUROPEAN SATELLITES

In figure 8 a few experimental data concerning the damping ratio of European satellites are reported; the values of the damping ratio are plotted vs input level. More informations are available in Ref. 7 and their analysis allows the following observations:

- higher damping values are determined at higher input level
- lower damping values are obtained at lower-frequency
- in general damping ratio varies from 0.03 to 0.066 for input levels rancing from 0.5 to 2.

CONCLUDING REMARKS

Several experiments have been carried out on aerospace structures to study their damping properties, but they do not appear sufficient to make reliable predictions for new structures. Furthermore the interference between two modes may contribute to misleading informations. It is extremely difficult to make comparisons between the data of two structural models because of various factors, such as different types of construction, different input levels, different test procedures.

It should be pointed out that overall damping depends on all the structural components, on the design philosophy and on the materials.

The validity of test equipment and procedures normally used have been discussed in detail for years, since they do not always appear to be the most suitable ones for a correct determination of damping.

In particular it seems to be highly recommended an accurate investigation into the effects of the following items on the value of damping:

- input level
- shaker size
- sweep rate
- recording equipment.

REFERENCES

1. Hay, J.A., *Experimentally Determined Damping Factors*, AGARD–CP–113, May 1973
2. Lazan, B.J., *Damping of Materials and Members in Structural Mechanics*, Pergamon Press, London 1968.
3. Gauzy, H., *Measurement of Inertial and Structural Damping*, AGARD Manual of Aeroelasticity, Vol. IV, Chapt. 3
4. Kellerer, K, Herkert, C.M., Mathes, H., *Composites in the Aerospace Industry: Current Applications and Future Developments*, Ispra Courses ACM/16/76, 1976.
5. Barboni, R., Nappi, A., Peroni, I., *Composite Materials for Space Applications*, presented at the 44 th AGARD–SMP Meeting, Lisbon, April 1977.
6. Barboni, R., Balis Crema, L., *Determinazione sperimentale delle caratteristiche viscoelastiche*, Aerotecnica, n. 2, 1972.
7. Carrington, H.G., Ottens, H.H., *A Survey of Data on Dampir, in Spacecraft Structures*, ESRO CR–539.
8. Poizat, Vialatoux, Vedrenne, Cochery, *Ammortissement viscoélastique*, L'Aéronatique et l'Astronautique, n. 58, 1976-3.
9. Kana, D.D., Huzar, S., *Synthesis of Shuttle Vehicle Damping Using Substructure Test Results*, J. Spacecraft, vol. 10, no. 12, Dec. 1973.
10. Kana, D.D., Unruh, J.F., *Substructure Energy Method for Prediction of Space Shuttle Model Damping*, J. Spacecraft, vol. 12, no. 5, May 1975.
11. Koenig, H.J., Drain, D.D., *Method of Relating Modal Damping to Local Dampers in Lumped–Parameter Systems*, NASA TN D–3637, Dec. 1966.

ACKNOWLEDGEMENTS

The Authors wish to thank Prof. Paolo Santini for promoting and supporting the present work at the "Istituto di Tecnologia Aerospaziale dell'Università di Roma".

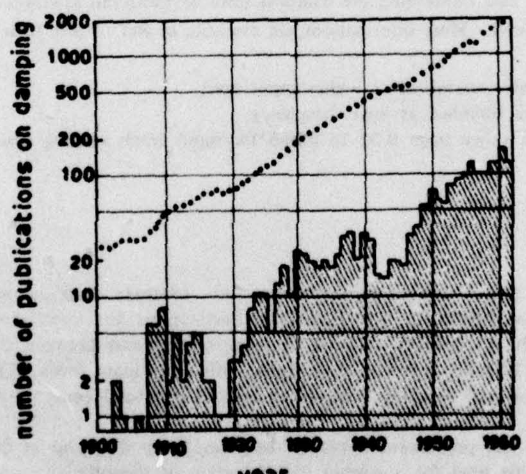

Fig. 1[2]– Number of Publications on Damping up to the 60'S

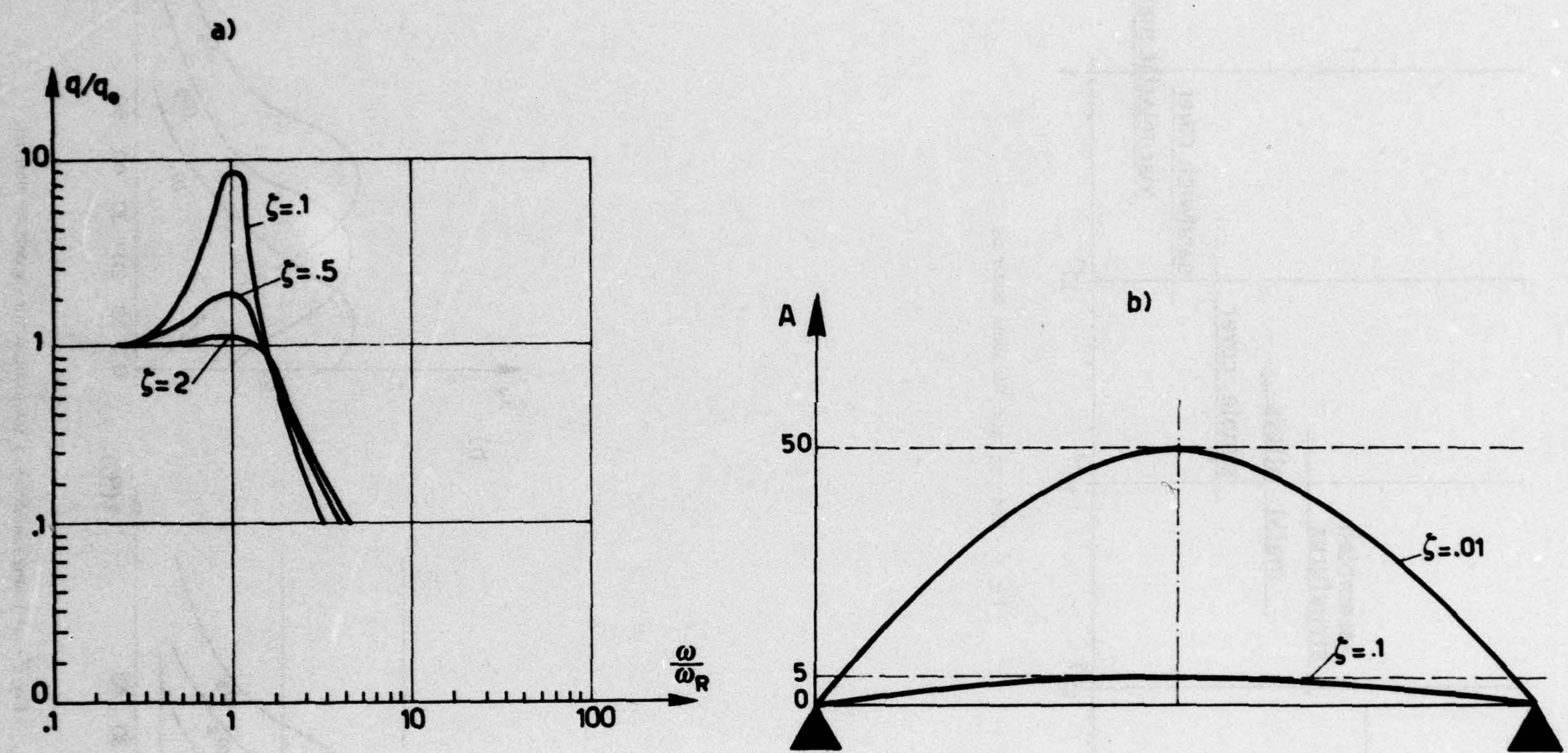

Fig. 2 – a) Influence of hysteretic damping on the response of a structure
b) Maximum deflection of a beam for different values of ζ

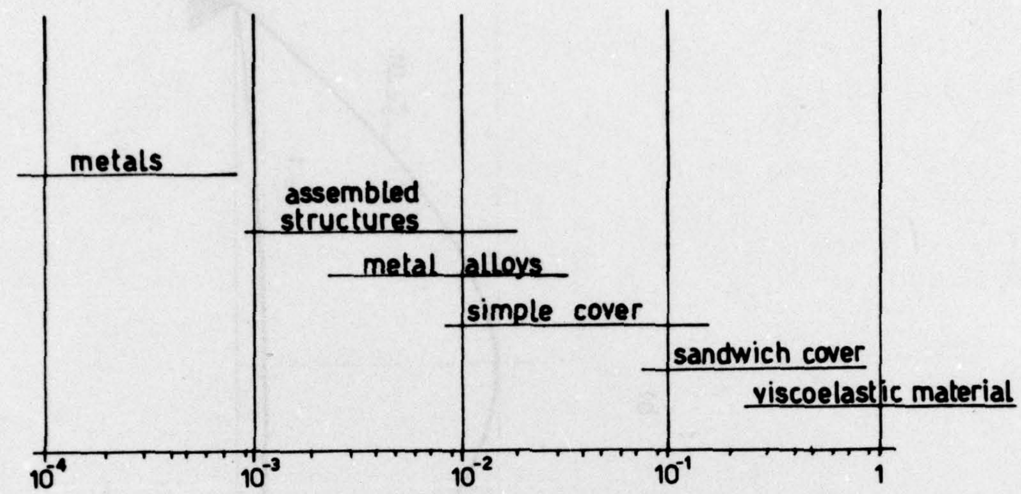

Fig. 3[a] – Damping ratio for some materials

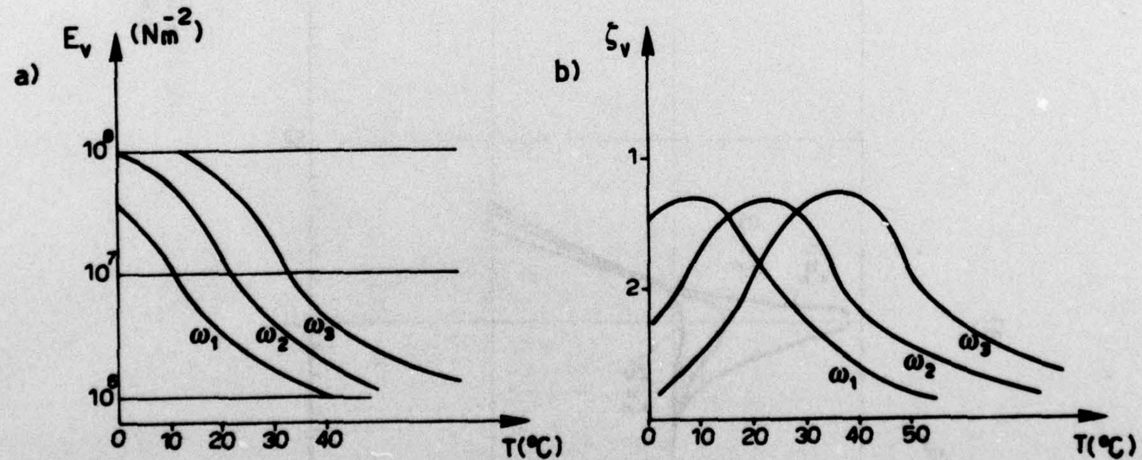

Fig. 4[a] – a) Young's modulus vs. temperature for viscoelastic materials
b) Damping ratio vs. temperature for viscoelastic materials

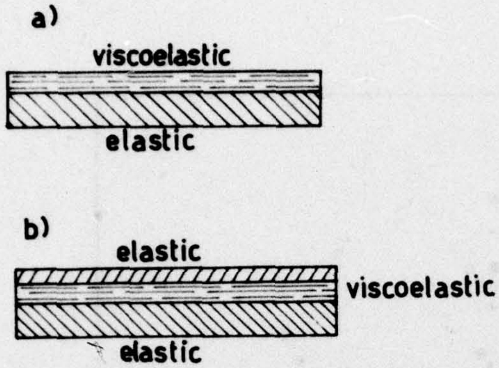

Fig. 5[8] – a) Simple cover
b) Multiple cover

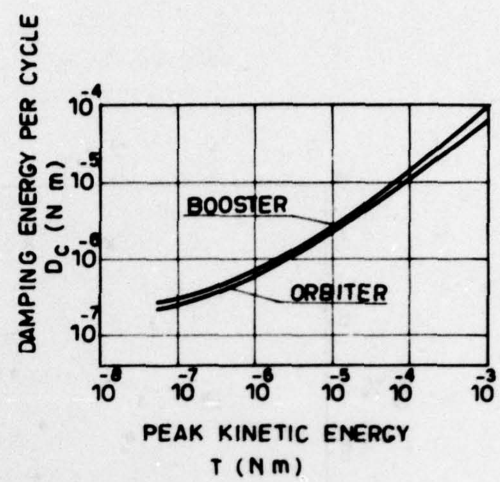

Fig. 6[9] – Experimental damping energy for free-free component configuration

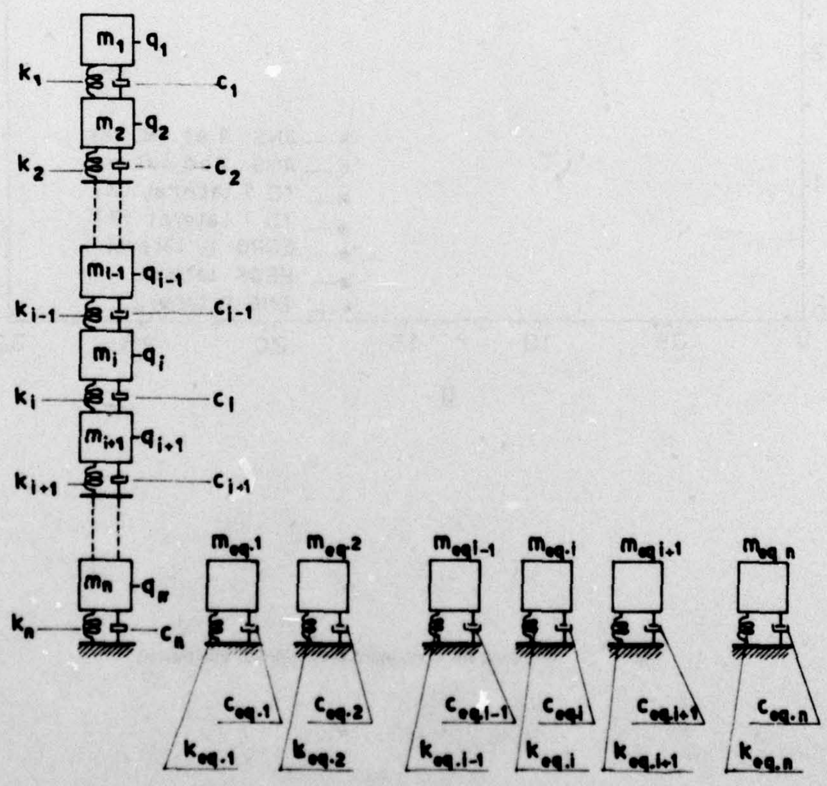

Fig. 7[11] – Lumped-parameter model and modal equivalent models

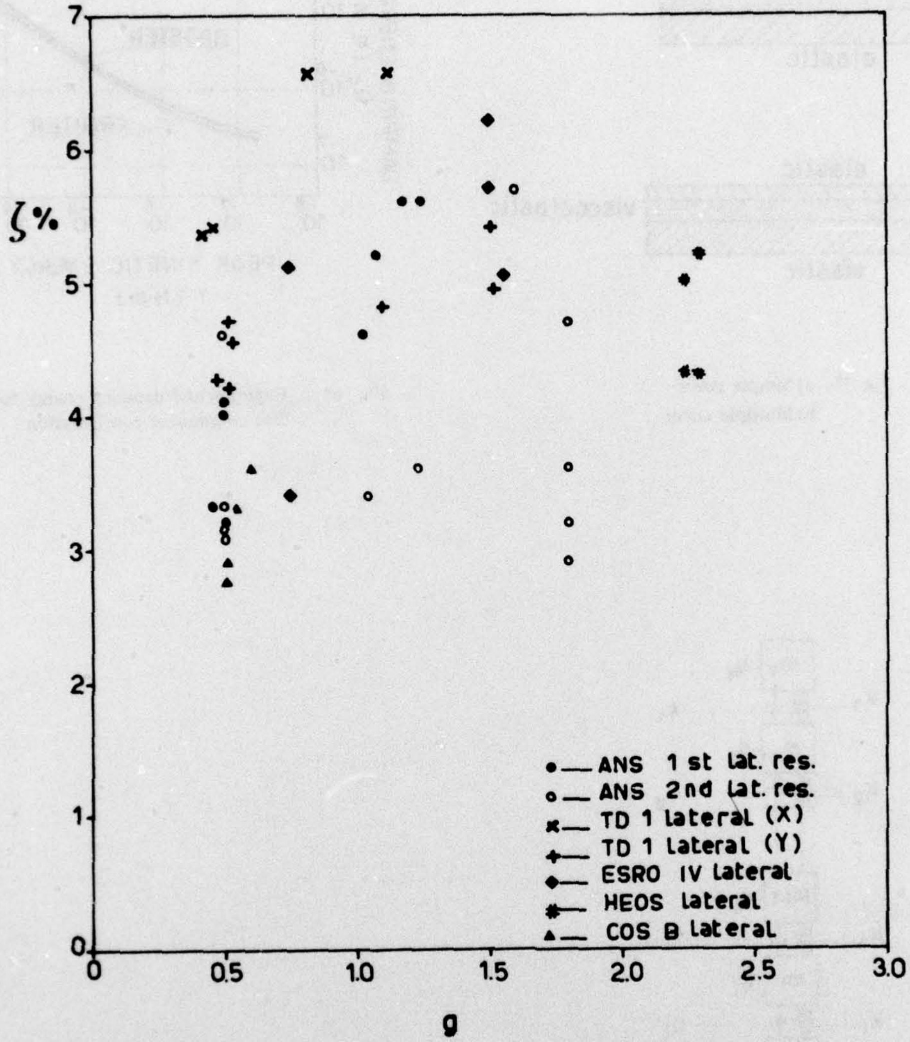

Fig. 8[7] – Damping ratio vs. input level (lateral excitation)

REPORT DOCUMENTATION PAGE

1. Recipient's Reference	2. Originator's Reference	3. Further Reference	4. Security Classification of Document
	AGARD-R-663	ISBN 92-835-1268-5	UNCLASSIFIED

5. Originator Advisory Group for Aerospace Research and Development
North Atlantic Treaty Organization
7 rue Ancelle, 92200 Neuilly sur Seine, France

6. Title

AN INTRODUCTION TO THE PROBLEM OF DYNAMIC STRUCTURAL DAMPING

7. Presented at the 45th Structures and Materials Panel Meeting,
Voss, Norway, September 1977.

8. Author(s)	9. Date
P.Santini, A.Castellani and A.Nappi	January 1978

10. Author's Address		11. Pages
	Istituto di Tecnologia Aerospaziale dell'Università di Roma Via Eudossiana 16, 00184 Roma, Italy	24

12. Distribution Statement This document is distributed in accordance with AGARD policies and regulations, which are outlined on the Outside Back Covers of all AGARD publications.

13. Keywords/Descriptors

Dynamic structural analysis	Airframes	Damping tests
Damping	Damping capacity	
Design criteria	Mathematical modes	

14. Abstract

In the design of aerospace structures, damping plays a fundamental role. Lack of knowledge of the damping ratio can result in either serious damage to the structure or over-dimensioned structure. The problem of determination of damping ratios is not yet resolved.

This Report indicates some fundamental aspects of the problem:

- the fields where damping is crucial,
- the types of structure involved,
- materials,
- mathematical simulation,
- test methods.

A bibliographic survey of numerical values completes the Report.

AGARD Report No.663
Advisory Group for Aerospace Research and Development, NATO
AN INTRODUCTION TO THE PROBLEM OF DYNAMIC STRUCTURAL DAMPING
P.Santini, A.Castellani and A.Nappi
Published January 1978
24 pages

In the design of aerospace structures, damping plays a fundamental role. Lack of knowledge of the damping ratio can result in either serious damage to the structure or over-dimensioned structure. The problem of determination of damping ratios is not yet resolved.

P.T.O.

AGARD-R-663

Dynamic structural analysis
Damping
Design criteria
Airframes
Damping capacity
Mathematical modes
Damping tests

AGARD Report No.663
Advisory Group for Aerospace Research and Development, NATO
AN INTRODUCTION TO THE PROBLEM OF DYNAMIC STRUCTURAL DAMPING
P.Santini, A.Castellani and A.Nappi
Published January 1978
24 pages

In the design of aerospace structures, damping plays a fundamental role. Lack of knowledge of the damping ratio can result in either serious damage to the structure or over-dimensioned structure. The problem of determination of damping ratios is not yet resolved.

P.T.O.

AGARD-R-663

Dynamic structural analysis
Damping
Design criteria
Airframes
Damping capacity
Mathematical modes
Damping tests

AGARD Report No.663
Advisory Group for Aerospace Research and Development, NATO
AN INTRODUCTION TO THE PROBLEM OF DYNAMIC STRUCTURAL DAMPING
P.Santini, A.Castellani and A.Nappi
Published January 1978
24 pages

In the design of aerospace structures, damping plays a fundamental role. Lack of knowledge of the damping ratio can result in either serious damage to the structure or over-dimensioned structure. The problem of determination of damping ratios is not yet resolved.

P.T.O.

AGARD-R-663

Dynamic structural analysis
Damping
Design criteria
Airframes
Damping capacity
Mathematical modes
Damping tests

AGARD Report No.663
Advisory Group for Aerospace Research and Development, NATO
AN INTRODUCTION TO THE PROBLEM OF DYNAMIC STRUCTURAL DAMPING
P.Santini, A.Castellani and A.Nappi
Published January 1978
24 pages

In the design of aerospace structures, damping plays a fundamental role. Lack of knowledge of the damping ratio can result in either serious damage to the structure or over-dimensioned structure. The problem of determination of damping ratios is not yet resolved.

P.T.O.

AGARD-R-663

Dynamic structural analysis
Damping
Design criteria
Airframes
Damping capacity
Mathematical modes
Damping tests

This Report indicates some fundamental aspects of the problem:

- the fields where damping is crucial,
- the types of structure involved,
- materials,
- mathematical simulation,
- test methods.

A bibliographic survey of numerical values completes the Report.

Papers presented at the 45th Structures and Materials Panel Meeting, Voss, Norway, September 1977.

ISBN 92-835-1268-5

This Report indicates some fundamental aspects of the problem:

- the fields where damping is crucial,
- the types of structure involved,
- materials,
- mathematical simulation,
- test methods.

A bibliographic survey of numerical values completes the Report.

Papers presented at the 45th Structures and Materials Panel Meeting, Voss, Norway, September 1977.

ISBN 92-835-1268-5

This Report indicates some fundamental aspects of the problem:

- the fields where damping is crucial,
- the types of structure involved,
- materials,
- mathematical simulation,
- test methods.

A bibliographic survey of numerical values completes the Report.

Papers presented at the 45th Structures and Materials Panel Meeting, Voss, Norway, September 1977.

ISBN 92-835-1268-5

This Report indicates some fundamental aspects of the problem:

- the fields where damping is crucial,
- the types of structure involved,
- materials,
- mathematical simulation,
- test methods.

A bibliographic survey of numerical values completes the Report.

Papers presented at the 45th Structures and Materials Panel Meeting, Voss, Norway, September 1977.

ISBN 92-835-1268-5